

# Optimal design of multi-conditions for axial flow pump

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**Abstract.** Passage components of the pump device will have a negative flow state when axial pump run off the design condition. Combined with model tests of axial flow pump, this paper use numerical simulation and numerical optimization techniques, and change geometric design parameters of the impeller to optimal design of multi conditions for Axial Flow Pump, in order to improve the efficiency of non-design conditions, broad the high efficient district and reduce operating cost. The results show that, efficiency curve of optimized significantly wider than the initial one without optimization. The efficiency of low flow working point increased by about 2.6%, the designed working point increased by about 0.5%, and the high flow working point increased the most, about 7.4%. The change range of head is small, so all working point can meet the operational requirements. That will greatly reduce operating costs and shorten the period of optimal design. This paper adopted the CFD simulation as the subject analysis, combined with experiment study, instead of artificial way of optimization design with experience, which proves the reliability and efficiency of the optimization design of multi-operation conditions of axial-flow pump device.

## 1. Introduction

Blade is the most important part of axial flow pump impeller, so the design quality of the blade directly determines the performance of the pump. The common methods of the design of axial flow pump blade are lifting method, circular arc method and singularity distribution method. The optimization design method based on CFD has been developed rapidly with the rapid development of computer technology. According to the numerical calculation results of axial flow pump, adjust the geometrical parameters of the axial flow pump blade, so that flow pattern in pump is better and the whirlpool, reflux and secondary flow and other unstable flow can be avoided[1–2]. At present, optimum design methods for axial flow impeller are all just single objective or multi objective optimization design for axial flow pump design working conditions, and there is nobody does hydraulic performance optimization design of multi-conditions[3–6]. Wang Kai, etc.[7] and Yuan Shouqi, etc.[8] from Jiangsu University studied the optimal design of multi conditions for centrifugal



pump. They designed an impeller with good external characteristics and the distribution of the velocity field, pressure field and so on more reasonable. Some other scholars in China also studied the optimal design of multi-conditions of wind turbine blades, turbine and turbine blades [9–11]. However, the head of axial flow pump for large pumping station is determined by different seasons and the level difference, so it is not in operation at the design working conditions all the time. In contrast, most of the time, the head is in non-design conditions, even a long part of the time it's in the highest or lowest running conditions. That will lead to a big increase in operating costs. It is very important to improve the operating efficiency of the non design conditions, expand the scope of the high efficient area and reduce the operating cost.

The flow of the liquid flow through the axial flow pump in designed conditions is considered to be in the best condition, which is similar to the ideal flow. When deviate from the design conditions, due to the liquid's viscosity, it will produce vortex, reflux, stall and flow in the pump, and these undesirable flow patterns will gradually increase. Change the geometric design parameters of axial flow pump impellers, reduce the adverse flow within the non design conditions, improve the efficiency of non design conditions, and make the design operating point efficiency to maintain a high is the main research contents of this paper. Combined with an axial flow pump model test, this paper used numerical simulation method and numerical optimization techniques, selected parameters of the impeller appropriately. The axial flow pump model is optimized to improve the efficiency of the pumping unit, and to broaden the range of the high efficiency zone of the axial flow pump, which provides a reference for the optimal design of axial flow pump.

## **2. Parametric Model of Axial Flow Impeller**

Under normal circumstances, in the design of the impeller, the blade is equally divided into 11 airfoil sections in the radial direction. In this paper, the basic design parameters of the axial flow impeller are entered into program written in FORTRAN to generate three-dimensional coordinates of the 11 section of the airfoil. The coordinate value file is imported into Turbo-Grid, and the axial flow pump is modeled and divided into grid. In this paper, by changing cascade dense degree and the angle of the airfoil of each section to change the axial flow blade shape to study the hydraulic performance of the pump unit under small flow, design and large flow rate. So as to achieve the purpose of broadening the range of high efficiency and reducing the operation cost.

### *2.1. Cascade dense degree ( $l/t$ )*

Cascade dense degree is important geometric parameters of axial flow impeller. It directly affects the efficiency of the pump, and important parameter determines the pump cavitations performance. Cascade dense degree is determined by the terms of the minimum energy loss in cascade and good cavitations performance. The total area of pump blade will reduce when cascade dense degree is reduced. The pressure difference between the working face and the back side of the blade is increased, and the cavitations performance becomes worse. However, the total area of blade is reduced, and the corresponding reduced hydraulic friction losses, blade efficiency can be improved. Airfoil schematic diagram of axial-flow blade is shown in figure 1.

The cascade dense degree data of 11 sections along the blade spanwise are required if in the design of axial flow impeller, there are 11 airfoil section. The calculations of these data follow the rule that

intensity distribution from the hub to the rim approximately equal. Therefore, it's only necessary to determine cascade dense degree of blade tip and root, and these can be achieved through the FORTRAN programming. Design variables are reduced from 11 to 2, so the efficiency of blade design is improved. The blade tip cascade dense degree named the design parameter  $a_1$  and the blade root time cascade dense degree named the design parameter  $a_2$ .

$$a_1 = (l/t)_{\text{tip}} \quad (1)$$

$$a_2 = (l/t)_{\text{root}} / (l/t)_{\text{tip}} \quad (2)$$

$$t = 2\pi r/z \quad (3)$$

where:  $l$  is the correspond airfoil chord;  $t$  is the airfoil pitch;  $r$  is the relative radius of each section;  $z$  is the blade number.

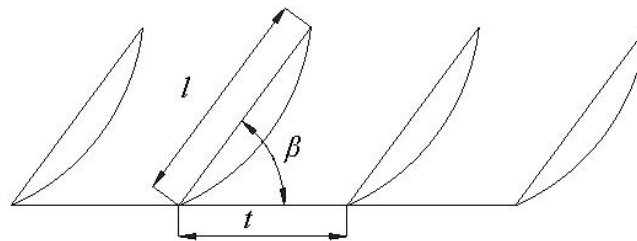


Fig.1 Airfoil schematic diagram of axial-flow blade

## 2.2. Airfoil angle( $\beta$ )

Airfoil angle of the blade has a very important influence on the performance of the axial flow pump. Generally, the outer edge airfoil of axial flow impeller vane is thin, almost in straight and the incidence of blade is small. Obviously, it's work capacity is not strong. However, if the airfoil of hub side is thick, the degree of crown is large and the incidence of blade is large, that will cause severe blade distortion. Therefore, the design should be targeted appropriately reduced the airfoil angle at the hub, reduce the hub side axial velocity and circumferential velocity component, while appropriately increase the airfoil angle of the outer edge, increase the incidence of the outer edge of blade, to improve the work capacity of the blade. This will not only reduce blade twist, improve working conditions of airfoil, increase the excess Flow, but also improve efficiency, expand effective area and improve the anti-cavitation performance of the blade. The airfoil angle is shown in figure 1.

In this paper, the initial design method for axial flow blade is based on CFD numerical calculation. Full 3-dimensional turbulent numerical simulation in the impeller of axial flow pump with different design parameters was calculated by CFX according to the design requirements. The numerical simulation results were analyzed and compared, taking into account the requirements of efficiency and cavitation performance, and get the final design result. Quadratic polynomial fitting these 11 data of airfoil angle, the results showed that the standard deviation was 0.999, the error is small. The relationship curve between airfoil angle and relative radius of each section along blade is as follows:

$$\beta = 90.504 - 129.96.4r + 57.26r^2 \quad (4)$$

Where:  $\beta$  for the airfoil angle;  $r$  for the relative radius of each section

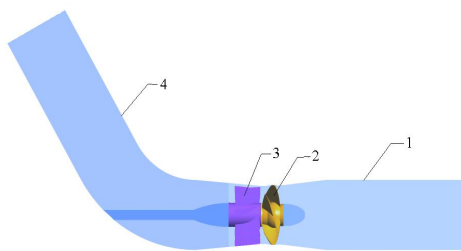
For each hub ratio, the relative radius of each airfoil is determined, so three coefficient values of the quadratic polynomial (4) can be changed by FORTRAN programming, then change the value of airfoil angle of each section. Define this 3 quadratic polynomial coefficients as  $a_3$ ,  $a_4$  and  $a_5$ , as the design variables of optimization design.

In the optimization design, changing the value of the above 5 variables can change the twist shape of the axial flow pump blade, then change the hydraulic performance of the axial flow pump device, improve the efficiency of the optimization and shorten the design cycle.

### 3. Numerical Simulation of Pump Device

#### 3.1. The establishment of calculation model

Axial flow device includes: a water inlet straight pipe section with a water guiding cone, an axial flow pump impeller, a guide vane and a standard 60 degree water outlet pipe section. In order to facilitate the modeling and meshing, the water guiding cone and the straight pipe section are integrated, and the water guiding cone is arranged into a stationary wall to have no influence on the flow pattern of the wheel hub area. In this paper, the nominal specific speed of axial flow impeller  $n_s = 800$ , design flow  $Q = 360$  L/s, design head  $H = 6.0$  m, rotating speed  $n = 1450$  r/min, and the blade tip unilateral gap is 0.2 mm. The rear guide vane is designed according to the design conditions of the impeller, diffusion angle of the guide vane is  $6^\circ$ , blade number of guide vane is 7, and the number of impeller blades is 4. Proe modeling is adopted in the inlet and outlet straight pipe elbow section, and the impeller and the guide vane are modeled by Turbo-Grid according to their 3D coordinate data points. Three dimensional numerical calculation model of axial flow pump device is shown in figure 2.



1.Inflow pipe 2. Impeller 3.Guide vane 4.Outflow pipe

Fig.2 Pump device numerical calculation model

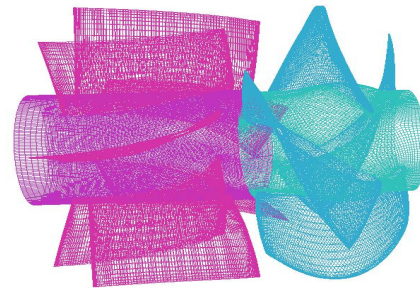


Fig.3 Impeller and guide vane grid chart

#### 3.2. Meshing

In this paper, ICEM software is used to mesh the 2 calculation domains of the inlet straight pipe section and the outlet Pipe Bend. The grid quality is above 0.4, and the quality is good, which meets the requirements of calculation. Structural mesh division of axial flow pump impeller and guide vane is carried out directly in Turbo-Grid, upon examination, the mesh quality of the impeller and the guide vane is good in Turbo-Grid, and at the same time be able to meet the requirements of orthogonality. When analyzing the grid independence, constantly changing the number of the grid and calculating the external characteristics of the pump device, found that when the grid increased to a certain amount, the efficiency of the pump device is stable. Under the condition of satisfying the requirements of the grid independence, the number of the impeller of the pump device is 330928, the number of the guide vane is 365274, and the whole computational domain mesh number is 1215277. The impeller and guide vane grid chart shown in figure 3.

### 3.3. Boundary condition

The input port of pump device computational domain is the input port of inlet pipe, inlet boundary condition is set to total pressure condition, that is the total pressure is set at the inlet to a standard atmospheric pressure. The output port of pump device computational domain is the output port of outlet Pipe Bend, outlet boundary condition is set to mass flow outlet, that is,  $Q$  is 360 kg/s. Impeller is set to rotate domains, in which the impeller rim wall boundary is set to reverse with respect to the impeller rotation speed, the rest of the computational domain are still. Impeller speed is 1450r / min. The blade surface, the outer edge of the hub, the inner rim surface and the other solid wall boundary conditions are using conditions that the solid wall surface meets the non slip of the viscous fluid, near-wall region using standard wall boundary conditions. The static and dynamic interface of water guide cone outlet ,the impeller inlet, outlet and the guide vane outlet using the average speed of the Stage model, quietly interface using the interface type None.

## 4. Experimental Verification

Use the standard k-e model, selected eight operating conditions of the axial flow pump device for numerical calculation. The calculation results are shown in table 1.

Table.1 Pump device numerical simulation results

$Q(L/S)$	$H(m)$	$\eta(\%)$	$Q(L/S)$	$H(m)$	$\eta(\%)$
280	8.397	0.7295	360	6.142	0.8467
300	7.735	0.7515	380	5.29	0.8419
320	6.9497	0.7663	400	4.15	0.7763
340	6.7	0.82	420	2.82	0.6376

According to table 1, when the flow rate is 360 L/s, the efficiency of the axial flow pump device is the highest, which is basically in accordance with the design requirements. The impeller, the corresponding guide vanes and the water inlet and outlet pipe are processed, and external characteristic experimental verification of the pump device were carried out in the high-precision test hall. The test conditions include the length of the inlet and outlet pipes, the position of the pressure pipe are strictly consistent with the numerical simulation, in order to ensure comparability of the test results. The predicted energy performance curve of the axial flow pump device is compared with the test results of the physical model, and the results are shown in figure 4.

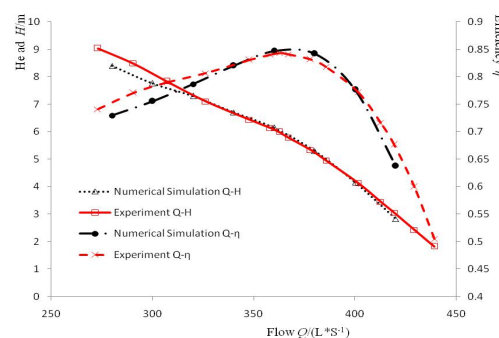


Fig.4 Comparasion of simulation results and experiment results

The figure shows that the predicted performance curve of the numerical simulation is consistent with the change trend of the experimental curve, the curve is in good agreement, and the error is less

than 3%. The accuracy and reliability of the numerical calculation of the axial flow pump device are demonstrated.

### 5. Multi condition optimization design of axial flow pump device

Using CFX numerical analysis software and Isight[12–16] numerical optimization software to optimize the design of axial flow pump system. According to  $Q$ - $H$  vane pump performance curves partition principle of vane pump proposed by Newman[1], The selection conditions were design conditions  $Q = 360$  L/s, low flow conditions  $Q = 300$  L/s, and high flow conditions  $Q = 420$  L/s. The efficiency is the highest in the design condition is the most important operating conditions in the engineering application. The efficiency of low flow conditions and high flow conditions is about 0.8 times and 1.2 times of the design condition.

#### 5.1. The establishment of optimization model

The purpose of optimization is, within the range of axial flow impeller design variables, under the constraints, to find the optimal values of design parameters, so that the efficiency of the three working conditions of the axial flow pump device is optimal. In this paper, according to the above calculation results, the multi-condition optimization problem of axial flow pump is defined as: the head change in small range under 3 flow conditions, constantly changing the value of the design variables, so that the efficiency of the three flow operating point has reached the optimal value, in order to broaden the efficient district range of axial flow device. In this paper, the above-mentioned axial flow pump device is the initial scheme, corresponding to the initial design variables of the impeller is:  $a_1 = 0.9885$ ,  $a_2 = 1.2897$ ,  $a_3 = 90.504$ ,  $a_4 = -129.96$ ,  $a_5 = 57.26$ . The optimization model as follows:

The objective function:

$$\max \eta(x) = w_1 \eta_1(x) + w_2 \eta_2(x) + w_3 \eta_3(x) \quad (5)$$

$$\text{Design points range: } \begin{cases} 0.85 \leq a_1 \leq 1.15 \\ 1.05 \leq a_2 \leq 1.45 \\ 88.504 \leq a_3 \leq 92.504 \\ -133.96 \leq a_4 \leq -125.96 \\ 49.26 \leq a_5 \leq 65.26 \end{cases} \quad (6)$$

$$\text{Constraints: } \begin{cases} 7.4 \leq H_1 \leq 8.0 \\ 6.0 \leq H_2 \leq 6.2 \\ 2.6 \leq H_3 \leq 3.2 \end{cases} \quad (7)$$

Design variables:  $x = [a_1, a_2, a_3, a_4, a_5]^T$

Where,  $\eta_1$ 、 $\eta_2$  and  $\eta_3$  are respectively the efficiency of the low flow conditions, design conditions and high flow conditions.  $w_1$ 、 $w_2$  and  $w_3$  are the corresponding weight value, and they are determined according to the actual running time of the pumping station at each working point. For the convenience of research, take  $w_1=0.3$ 、 $w_2=0.4$  and  $w_3=0.3$  in this paper.  $H_1$ 、 $H_2$  and  $H_3$  are respectively the head of each working condition point, the unit is meters. In order to ensure that the design point and specific speed of the axial flow pump impeller is unchanged after the optimization design, make the variation range of the head of design point as small as possible, the other two operating point head can vary slightly larger. Refer to the relevant reference to select the range of design variables[17].

### 5.2. Selection of optimization algorithm

For hydraulic performance conditions optimal design of multi-axial flow pump which is constrained, nonlinear, multi-target and solution is not only, this paper use Sequential Quadratic Programming(SQP), which is a kind of gradient optimization algorithm. The method can deal directly with equality and inequality constraints, it is recognized as one of the best methods to solve the nonlinear problem. It has good global convergence and local superlinear convergence properties, less iterations, fast convergence speed, with a strong rope border closing ability, so it is particularly suitable for the optimal design in this paper that the design variables are few, and the constraint conditions are not much.

### 5.3. The establishment of data flow

Isight is a parameter based multidisciplinary design optimization software, can be integrated simulation optimization software, to achieve a complete set of automatic optimization design calculation framework. the numerical simulation software CFX was used to analyze the hydraulic performance of the pump device. According to the design parameters, use the programs written by FORTRAN to generate three-dimensional coordinates of the impeller blades. Then impeller modeling and meshing in Turbo-grid according to the blade coordinates, each set of design variables will generate a new grid file of axial flow impeller. The grid of vane and inlet and outlet pipe are divided by Turbo-grid and ICEM respectively, then each part of the grid is introduced into the CFX to carry on the pre processing and numerical calculation of the 3 working conditions of the pump device. Each iteration is a complete set of the above calculation processing, where the iterative process of the grid of vane and inlet and outlet pipe are remained unchanged.

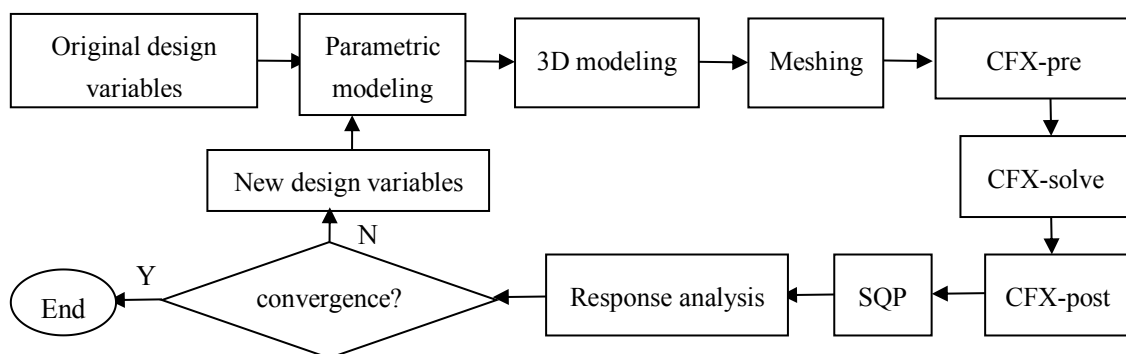


Fig.5 Flow chart of optimization design for axial flow pump device

### 5.4. Analysis of optimization result

The design variables of the axial flow pump impeller are constantly changed, so the total efficiency of the 3 working conditions of the axial flow pump device is the highest Within the head restraint range. After a month of continuous iterative calculation in workstation, get the final design of the impeller of the pump device. The optimization results are compared with the initial results, as shown in table 2.

Table.2 Pump device numerical optimization results

parameters	before optimization	before optimization	parameters	before optimization	before optimization
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$a_1$	0.9885	0.85	$H_1(\text{m})$	7.735	7.6457
$a_2$	1.2897	1.4468	$H_2(\text{m})$	6.142	6.0274
$a_3$	90.504	90.601	$H_3(\text{m})$	2.82	3.1996
$a_4$	-129.96	-129.84	$\eta_1$	0.7515	0.77685
$a_5$	57.26	57.278	$\eta_2$	0.84467	0.8495
—	—	—	$\eta_3$	0.6376	0.7102

According to the results of table 2, the blade tip cascade dense degree decreases, and the root time cascade dense degree increases, so the length difference of inside and outside airfoil is reduced, the head of blade outlet is balanced and the radial flow is reduced. The hydraulic performance of the impeller is improved. At the same time, it can be found that according to the change of the airfoil placed angle fitting coefficient, rim flank type setting angle increases, the hub flank type setting angle decreases, so the distortion of the impeller blade shape will be reduced. The working conditions of airfoil will be improved, which is consistent with the axial flow impeller design optimization ideas. Optimization results show that the efficiency of design condition has been improved, but the increase is not obvious, and the efficiency of high flow rate operating point and low flow rate condition improve more obvious, the high flow rate operating point increased by 7.4%, the low flow rate condition increased by 2.6%. The optimization effect is obvious.

The hydraulic performance of the rest working point is calculated by numerical simulation, and compared with the hydraulic performance before optimization, the results are shown in figure 6.

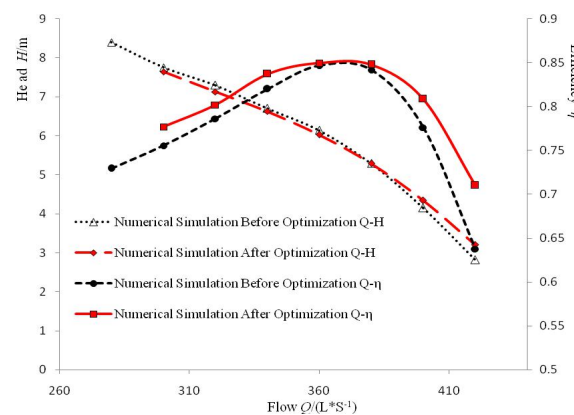


Fig.6 Pump device curves before and after optimization

According to performance curves of pump unit before and after optimization in figure 6, after optimization, the head of low flow conditions and design conditions slightly reduced, but the efficiency is improved; high flow condition head has increased, the efficiency is also improved. After optimization, the efficiency curves are all raised, efficient district range widened, so the operational stability of pumping station will be improved, and the operating cost of pumping station will be reduced, optimization effect is very obvious.

## 6. Conclusion

1) Based on the numerical analysis and numerical optimization technique, a complete set of method of optimum design of axial flow pump is presented. The method can greatly reduce the optimization design cost and shorten the design period.



2) Use disciplinary analysis approach of CFD calculations, combined with experimental research means to replace artificial empirically optimized methods to improve the reliability of the optimization results, and it also proves the reliability and efficiency of the multi condition optimization design of axial flow pump.

3) After optimization, the efficiency of the low flow working point of the axial flow pump device is increased by about 2.6%, and the design point efficiency is increased by about 0.5%, and the efficiency of the high flow working point is increased by up to 7.4%. The high efficiency district of the axial flow pump is obviously widened, which greatly reduces the operating cost of the pumping station. The optimization effect is very obvious.

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